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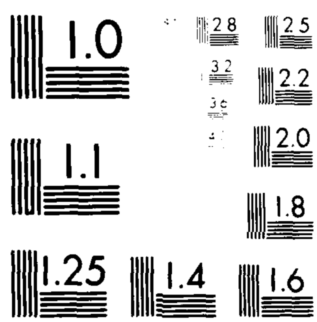

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TECHNICAL REPORT ARLCB-TR-82002

EFFECT OF SUPPORT CONDITIONS ON BEAM VIBRATIONS SUBJECTED TO MOVING LOADS

Julian J. Wu

January 1982

US ARMY ARMAMENT RESEARCH AND DEVELOPMENT COMMAND
LARGE CALIBER WEAPON SYSTEMS LABORATORY
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Solutions of beam vibrations under moving loads are presented with a variety of support conditions. The purpose is to demonstrate how the support conditions will effect such beam motions. The solution method and mathematical background will be reviewed including the introduction of various support parameters. By slightly modifying an existing computational scheme, the desired results have been obtained and presented in several tables and plots showing the effect of support stiffness on beam motions.

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STATEMENT OF THE PROBLEM

The dynamic equation of Euler-Bernoulli beam subjected to a moving force can be written as the following equation

$$EIy'''' + \rho A\ddot{y} = P\delta(x-\bar{x}) \quad , \quad \begin{matrix} 0 \le x \le \ell \\ 0 \le t \le T \end{matrix} \tag{1}$$

where $y = y(x,t)$ denotes the beam deflection as a function of the spatial coordinates x and the time t. The letters E, I, A, ℓ, and ρ denote elastic modulus, second moment of the cross-sectional area, and the area itself, the length, and the material density of the beam, respectively. A Dirac delta function is denoted by δ, $\bar{x} = \bar{x}(t)$ is the location of the force P. T denotes some finite time of interest. As usual, a prime (') denotes differentiation with respect to x and a dot (·), differentiation with respect to t.

The boundary conditions are written as

$$\begin{aligned} EIy'''(0,t) + k_1y(0,t) &= 0 \\ EIy''(0,t) - k_2y'(0,t) &= 0 \\ EIy'''(\ell,t) - k_3y(\ell,t) &= 0 \\ EIy''(\ell,t) + k_4y'(\ell,t) &= 0 \end{aligned} \tag{2}$$

where k_i, i = 1, 2, 3, and 4 are the spring constant which model the support characteristics. The initial conditions are

$$\begin{aligned} y(x,0) &= \bar{y}_0(x) \\ \dot{y}(x,t) &= \bar{y}_1(x) \end{aligned} \tag{3}$$

Equations (1) through (3) will be written in dimensionless form for the sake of generality and simplicity. This is accomplished by the introduction of dimensionless parameters. In the following an arrow (→) will be read as "replaces":

$$y \to \frac{y}{\ell} \quad , \quad x \to \frac{x}{\ell} \quad , \quad t \to \frac{t}{T}$$

$$k_1 \to \frac{k_1 \ell^3}{EI} \quad , \quad k_2 \to \frac{k_2 \ell}{EI}$$

$$k_3 \to \frac{k_3 \ell^3}{EI} \quad , \quad k_4 \to \frac{k_4 \ell}{EI}$$

$$y_1 \to T y_1 \quad , \quad P \to \frac{P \ell^3}{EI}$$

With these new dimensionless parameters, Eqs. (1) through (3) become

$$y'''' + \gamma^2 \ddot{y} = \bar{P}\delta(x-\bar{x}) \quad , \quad \begin{array}{l} 0 \leq x \leq 1 \\ 0 \leq t \leq 1 \end{array} \tag{1'}$$

$$\left.\begin{array}{l} y'''(0,t) + k_1 y(0,t) = 0 \\ y''(0,t) - k_2 y'(0,t) = 0 \\ y'''(1,t) - k_3 y(1,t) = 0 \\ y''(1,t) + k_4 y(1,t) = 0 \end{array}\right. \quad 0 \leq t \leq 1 \tag{2'}$$

and

$$\left.\begin{array}{l} y(x,0) = \bar{y}_0(x) \\ \dot{y}(x,0) = y_1(x) \end{array}\right. \quad 0 \leq x \leq 1 \tag{3'}$$

where in Eq. (1'), we have

$$\gamma = \frac{c}{T} \tag{5a}$$

which is dimensionless, with

$$c = \left(\frac{\rho A \ell^4}{EI}\right)^{1/2} \tag{5b}$$

which has the dimension of the physical time.

Hence then, we shall obtain solutions for the problem defined by Eqs. (1') through (3') for various values of k_i, $i = 1, 2, 3$, and 4.

AN EQUIVALENT VARIATIONAL PROBLEM

The problem of solving the equation (1′) through (3′) in the previous section will be transformed into a variational problem. Consider

$$\delta I = 0 \tag{5a'}$$

with

$$I = \int_0^1\int_0^1 \{[illegible]\, y^{*\prime\prime} - [illegible]\, \ddot{y}y^* + [illegible]\, \delta(\xi-\bar{x})y^*\}\, dxdt$$

$$+ \int_0^1 dt\, \{k_1 y(0,t)y^*(0,t) + k_2 y'(0,t)y^{*\prime}(0,t)$$

$$+ k_3 y(1,t)y^*(1,t) + k_4 y'(1,t)y^{*\prime}(1,t)\}$$

$$+ [illegible] \int_0^1 dx\, \{k_5[y(x,0) - \bar{y}_0(x)]y^*(x,1) - \bar{y}_1(x)y^*(x,0)\} \tag{5b'}$$

where $y^*(x,t)$ is called the adjoint function of $y(x,t)$. If one takes the first variation of I considering $y(x,t)$ to be fixed and δy^* to be completely arbitrary, it is easy to see that the differential equation (1′) and boundary condition (2′) will be recovered and the initial condition becomes

$$\dot{y}(x,1) - k_5[y(x,0) - \bar{y}_0(x)] = 0$$

$$\dot{y}(x,0) - \bar{y}_1(x) = 0 \tag{3''}$$

In Eq. (3″), it is seen that if one let $k_5 \to \infty$ in the limit, the initial condition of (3′) will also be recovered. The use of a large parameter such as k_5 is known as the penalty function method or the method of large spring constant.[1]

---

[1] Wu, J. J., "Vibrations of a Beam Under Moving Loads by a Finite Element Formulation Consistent in [illegible] and Time Coordinates," The 51st Shock and Vibration Bulletin, Part 3; Analytical Methods, Dynamics, and Vehicle Systems, pp. 111-130 (1981).

OUTLINE OF SOLUTION FORMULATION

To derive the finite element matrix equations, one begins [illegible]

and write

$$
\begin{aligned}
&[\text{illegible}] = \\
&= \int_0^1 \int_0^1 [y'' \delta v^{*''} - [\text{illegible}]\, \dot{v}\, \delta \dot{v}^{*} + [\text{illegible}]]\, [\text{illegible}] \\
&\quad + \int_0^1 dt\, [k_1 y(1,t)\, \delta v^{*}(1,t) + [\text{illegible}]] \\
&\quad + k_2 v(1,t)\, \delta v^{*}(1,t) + [\text{illegible}] \\
&\quad + [\text{illegible}]
\end{aligned}
$$

The element local variables are now introduced

$$\xi = \xi^{(i)} = Kx - i + 1$$

$$\eta = \eta^{(j)} = Lt - j + 1 \qquad (7)$$

or

$$x = \frac{1}{K} (\xi + i - 1)$$

$$t = \frac{1}{L} (\eta + j - 1)$$

where K is the number of divisions in x and L, in η. A typical grid scheme is shown in Figure 6). Equation (6b) can now be written as

$$\sum_{i=1}^{K}\sum_{j=1}^{L}\int_0^1\int_0^1 \left[\frac{K^3}{L}\, y''_{(ij)}\,\delta y^{*''}_{(ij)} - \frac{\gamma^2 L}{K}\, y_{(ij)}\,\delta y^{*}_{(ij)}\right] d\xi\, d\eta$$

$$+ \sum_{j=1}^{L}\int_0^1 d\eta \left\{\frac{K_1}{L}\, y_{(ij)}(0,\eta)\,\delta y^{*}_{(ij)}(0,\eta) + K_2\,\frac{K^2}{L}\, y'_{(ij)}(0,\eta)\,\delta y^{*'}_{(ij)}(0,\eta)\right\}$$

$$+ \sum_{i=1}^{K}\int_0^1 \frac{1}{K}\left\{\gamma^2 k_5\left(y_{(ij)}(\xi,1)\,\delta y^{*}_{(ij)}(\xi,1)\right)\right\}$$

$$= -\sum_{i=1}^{K}\sum_{j=1}^{L}\int_0^1\int_0^1 \delta(x-\bar{x})\,\delta y^{*}_{(ij)}(\xi,\eta)\, d\xi\, d\eta$$

$$+ \sum_{i=1}^{K}\frac{\gamma^2 k_5}{K}\int_0^1 d\xi\left\{\bar{y}_0(i)(\xi)\,\delta y^{*}_{(iL)}(\xi,1)\right\}$$

$$+ \sum_{i=1}^{K}\frac{\gamma^2}{K}\int_0^1 d\xi\left\{\bar{y}_1(i)\,\delta y^{*}_{(i1)}(\xi,0)\right\} \tag{8}$$

The shape function vector is now introduced. Let

$$y_{(ij)}(\xi,\eta) = \underset{\sim}{a}^T(\xi,\eta)\, Y_{(ij)}$$

$$y^{*}_{(ij)}(\xi,\eta) = \underset{\sim}{a}^T(\xi,\eta)\, \underset{\sim}{Y}^{*}_{(ij)} = \underset{\sim}{Y}^{*T}_{(ij)}\, \underset{\sim}{a}(\xi,\eta) \tag{9}$$

Equation (8) then becomes

$$\sum_{i=1}^{K}\sum_{j=1}^{L} \delta\underset{\sim}{Y}^{*T}(ij) \left\{\frac{K^3}{L}\underset{\sim}{A} - \frac{\gamma^2 L}{K}\underset{\sim}{B}\right\} \underset{\sim}{Y}(ij)$$

$$+ \sum_{i=1}^{L} \delta\underset{\sim}{Y}^{*T}(ij) \left\{\frac{k_1}{L}\underset{\sim}{B_1} + \frac{k_2 K^2}{L}\underset{\sim}{B_2}\right\} \underset{\sim}{Y}(ij)$$

$$+ \sum_{i=1}^{L} \delta\underset{\sim}{Y}^{*T}(Kj) \left\{\frac{k_3}{L}\underset{\sim}{B_3} + \frac{k_4 K^2}{L}\underset{\sim}{B_4}\right\} \underset{\sim}{Y}(ij)$$

$$+ \sum_{i=1}^{K} \delta\underset{\sim}{Y}^{*T}(iL) \left\{\frac{\gamma^2 k_5}{K}\underset{\sim}{B_5}\right\} \underset{\sim}{Y}(iL)$$

$$= \sum_{i=1}^{K}\sum_{j=1}^{L} \delta\underset{\sim}{Y}^{*T}(ij)\, \frac{p}{L}\, F(ij) + \sum_{i=1}^{K} \delta\underset{\sim}{Y}^{*T}(iL)\, \frac{\gamma^2 k_5}{K}\, \underset{\sim}{G}(i)$$

$$+ \sum_{i=1}^{K} \delta\underset{\sim}{Y}^{*T}(i1)\, \frac{\gamma^2}{K}\, \underset{\sim}{H}(i) \tag{10}$$

where, as it can be seen readily, that

$$\underset{\sim}{A} = \int_0^1 \int_0^1 \underset{\sim}{a}_{,\xi\xi}\, \underset{\sim}{a}^T_{,\xi\xi}\, d\xi d\eta$$

$$\underset{\sim}{B} = \int_0^1 \int_0^1 \underset{\sim}{a}_{,\eta}\, \underset{\sim}{a}^T_{,\eta}\, d\xi d\eta$$

$$\underset{\sim}{B_1} = \int_0^1 \underset{\sim}{a}(0,\eta)\underset{\sim}{a}^T(0,\eta)d\eta \quad , \quad \underset{\sim}{B_2} = \int_0^1 \underset{\sim}{a}_{,\xi}(0,\eta)\underset{\sim}{a}^T_{,\xi}(0,\eta)d\eta$$

$$\underset{\sim}{B_3} = \int_0^1 \underset{\sim}{a}(1,\eta)\underset{\sim}{a}^T(1,\eta)d\eta \quad , \quad \underset{\sim}{B_4} = \int_0^1 \underset{\sim}{a}_{,\xi}(1,\eta)\underset{\sim}{a}^T_{,\xi}(1,\eta)d\eta$$

$$\underset{\sim}{B_5} = \int_0^1 \underset{\sim}{a}(\xi,1)\underset{\sim}{a}^T(\xi,0)d\xi$$

$$\underset{\sim}{F}(ij) = \int_0^1 \int_0^1 a(\xi,\eta)\bar{\delta}_{(ij)}(\xi-\bar{\xi})d\xi d\eta \quad , \quad \underset{\sim}{G}(i) = \int_0^1 a(\xi,1)\bar{y}_{0(i)}(\xi)d\xi$$

and

$$\underset{\sim}{H}_i = \int_0^1 \underset{\sim}{a}(\xi,0)\bar{y}_{1(i)}(\xi)d\xi \tag{11}$$

Now Eq. (10) can be assembled in a global matrix equation

$$\delta Y^{*T} K Y = \delta Y^{*} F \qquad (12)$$

By virtue of the fact that δY^* is not subjected to any constrained conditions, one has

$$K Y = F \qquad (13)$$

which can be solved routinely. More details can be found in reference 1.

RESULTS AND DISCUSSIONS

Appropriate values of physical parameters must be assigned for obtaining numerical results. Let

$$v = \frac{\ell}{T} \qquad (14)$$

be the velocity of the travelling force. Only constant velocity will be considered in this report. Thus T becomes the time required for the force to move from one end of the beam to the other end. As T varies from ∞ to 0, the velocity v varies from 0 to ∞ as ℓ is always finite. Since we have normalized all the parameters in length with respect to ℓ, it is not necessary to specify ℓ in numerical computations. Instead, the beam's length is considered to be unity. The real value in length can be recovered simply by a multiplication of ℓ to the normalized (dimensionless) ones. Then it will be helpful if v in Eq. (14) can be related to some reference velocity associated with the beam's characteristics. We shall select the velocity of the first mode of vibration (standing waves) of a cantilevered beam as this reference velocity and call it

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[1] Wu, J. J., "Vibrations of a Beam Under Moving Loads by a Finite Element Formulation Consistent in Spatial and Time Coordinates," The 51st Shock and Vibration Bulletin, Part 3; Analytical Methods, Dynamics, and Vehicle Systems, pp. 111-130 (1981).

v_1. Hence the normalized velocity is defined by

$$\bar{v} = \frac{v}{v_1} \tag{15}$$

Now, we shall relate this $\bar{v}$ with the parameters defined earlier in this report. It is known in many textbooks on vibration (for example, see reference 3) that for a cantilevered beam, the first mode of vibrations has a circular frequency

$$\omega_1 = (1.875)^2/c = 3.516/c \tag{16}$$

where c is given in (5b). The corresponding frequency and period are then respectively

$$f_1 = \frac{\omega_1}{2\pi} = 0.560/c \quad \text{(in cycles per seconds)}$$

$$T_1 = \frac{1}{f_1} = 1.79 \quad \text{(in seconds)} \tag{16}$$

Hence

$$v_1 = \frac{2\ell}{T_1} = 2\ell f_1 = 1.12\frac{\ell}{c} \tag{17}$$

and

$$\bar{v} = \frac{v}{v_1} = \frac{\ell/T}{1.12\ \ell/c} = 0.895\frac{c}{T} = 0.895\gamma \tag{18}$$

where $\gamma = c/T$, as defined in Eq. (5b).

For the results computed in this report, γ is set to be ten. Hence $\bar{v}$ = 8.95 or v is about nine times v_1. At this load velocity, the dynamic effect of the moving force on the beam vibration is quite evident as shown in Figures 1 through 5. This is also approximately a typical speed at which a projectile

---

[3]Fryba, L. Vibration of Solids and Structures Under Moving Loads, Noordhoff International, Groningen, 1971, p. 91.

moves down a cannon tube (see, for example, Reference 4). Figure 1 shows the deflection curves for a beam with fixed-fixed supports. Curves numbered I, II, III, and IV are at the moment when P is at 1/4, 2/4, 3/4, and 4/4 of support. Figures 2 through 5 are the same curves for the support conditions of fixed-simply supported, fixed-free, free-free and free-fixed respectively.

It should be noted that the load P is assumed to move from the left toward the right. The beam motions from fixed-free supports are quite different from those of free-fixed supports as demonstrated, for example, in Figures 3 and 5. This dynamic behavior has also been noted by Fryba.[3] In fact, the deflection curves for the case of free-fixed supports resemble more closely to those of free-free supports than the fixed-free ones.

The values of the "spring constants" in Eqs. (2') and (5b) are as follows. For a fixed rigid support the k_i is taken to be 10^{10}; for no support at all, it is assigned a zero. The value of k_5 in Eq. (5b) is also assigned to be 10^{10}.

Results presented above are based on a grid scheme of 5x8 elements. (A typical grid scheme is shown in Figure 6). Numerical convergence of these data should be fairly good as discussed previously in Reference 1.

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[1]Wu, J. J., "Vibrations of a Beam Under Moving Loads by a Finite Element Formulation Consistent in Spatial and Time Coordinates," The 51st Shock and Vibration Bulletin, Part 3; Analytical Methods, Dynamics, and Vehicle Systems, pp. 111-130 (1981).

[3]Fryba, Vibration of Solids and Structures Under Moving Loads, Noordhoff International, Groningen, 1971, p. 91.

[4]Wu, J. J., "Vibrations of a Beam Under Moving Loads by a Finite Element Formulation Consistent in Spatial and Time Coordinates," The 51st Shock and Vibration Bulletin, Part 3; Analytical Methods, Dynamics, and Vehicle Systems, pp. 126 (1981).

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3. Fryba, L., Vibration of Solids and Structures Under Moving Loads, Noordhoff International, Groningen, 1971, p. 91.
4. Wu, J. J., "Vibrations of a Beam Under Moving Loads by a Finite Element Formulation Consistent in Spatial and Time Coordinates," The 51st Shock and Vibration Bulletin, Part 3; Analytical Methods, Dynamics, and Vehicle Systems, pp. 126 (1981).

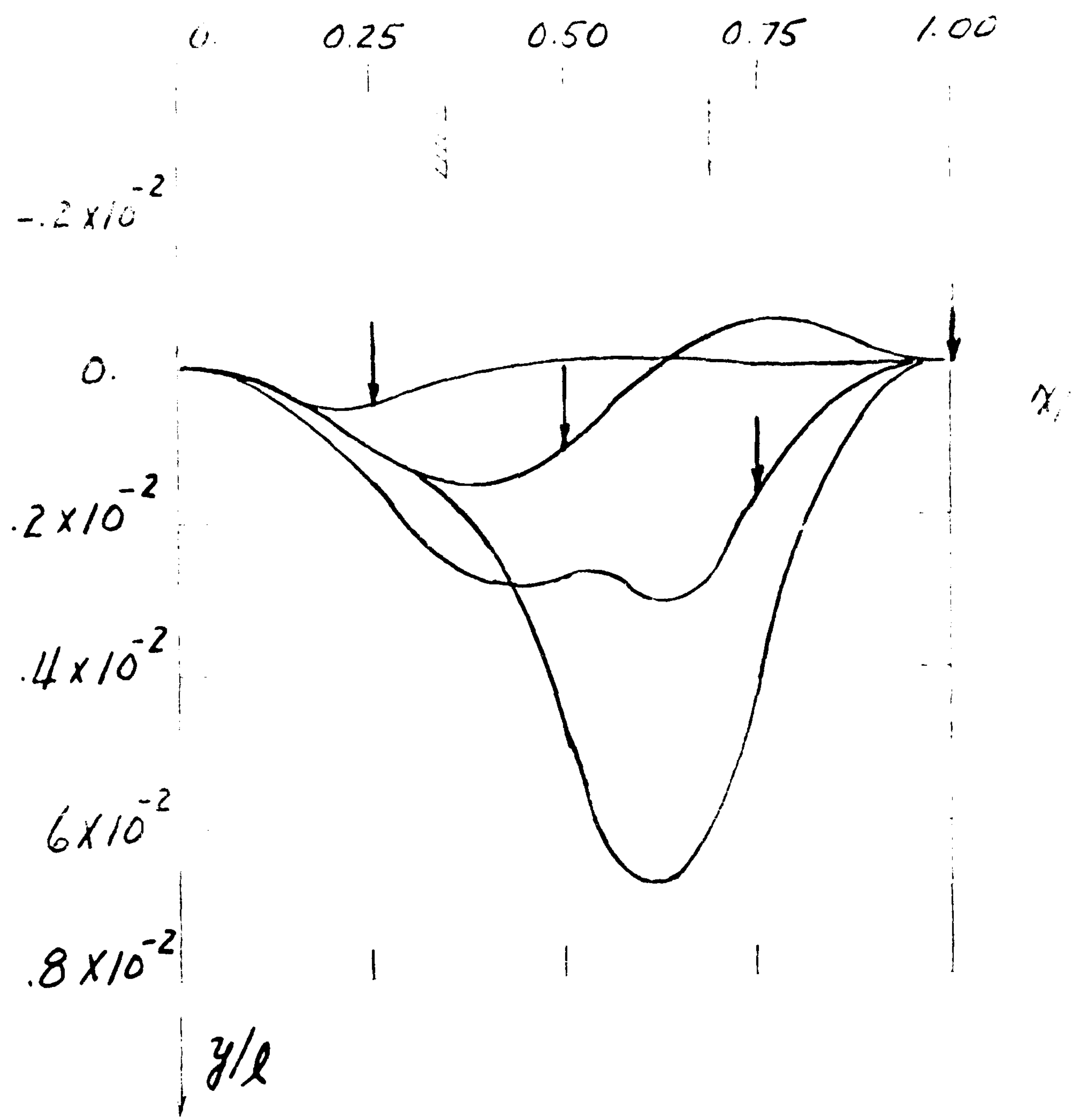

Figure 1. Beam deflections under a moving load for fixed-fixed supports (γ = 10.0).

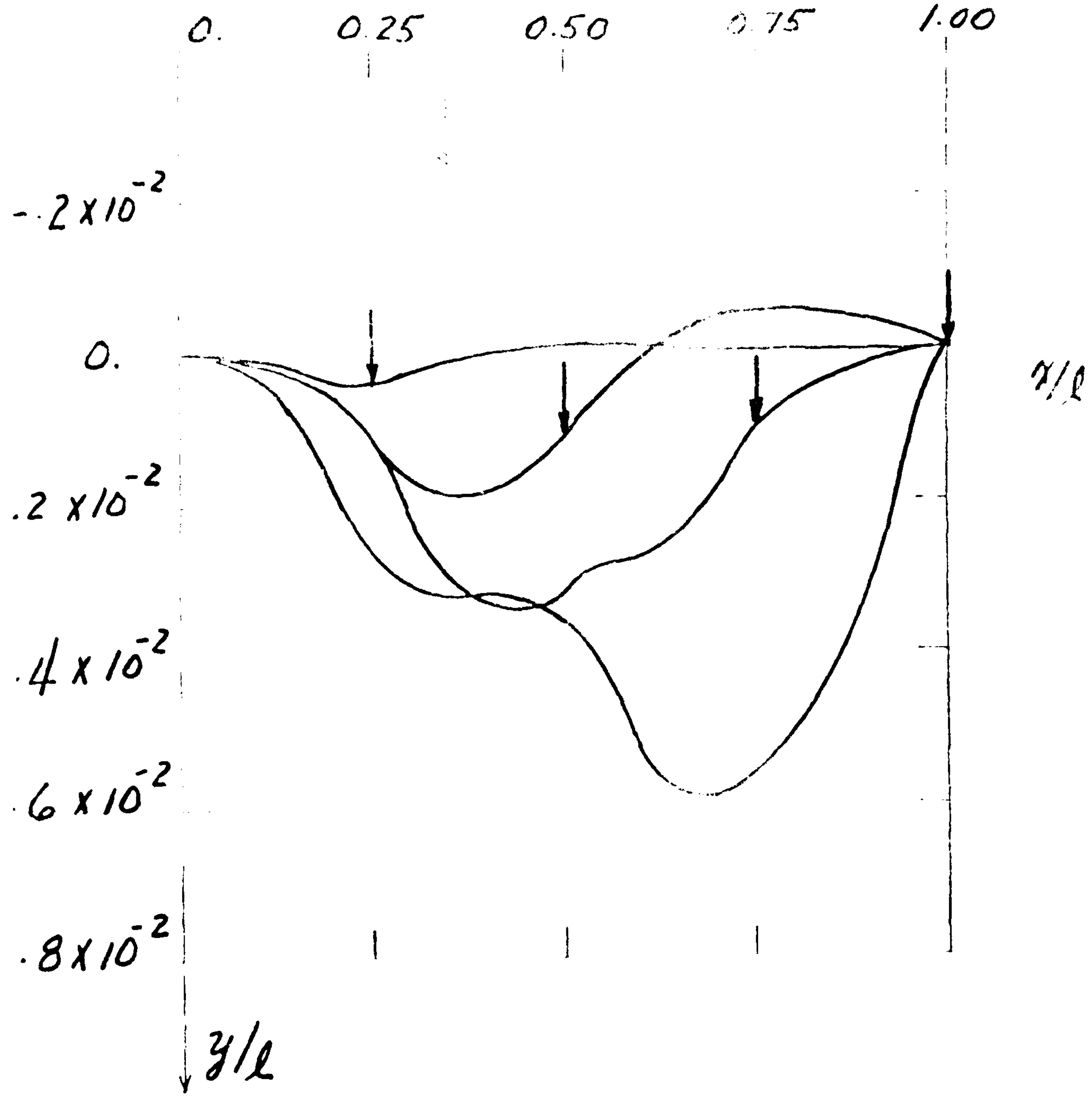

Figure 2. Beam deflections under a moving load for fixed-simply supported end conditions ([illegible] = 10.0).

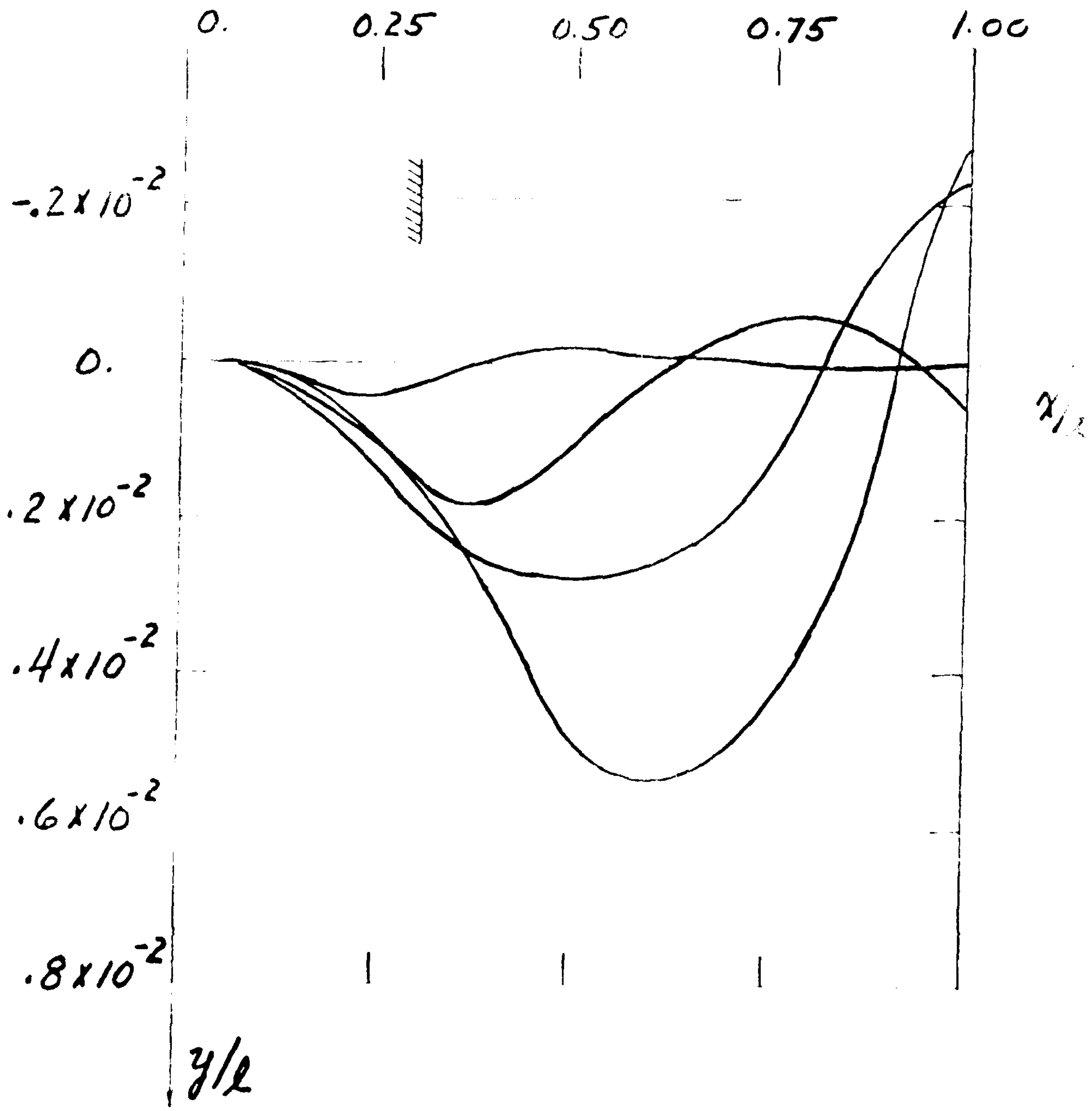

Figure 3. Beam deflections under a moving load for fixed-free supports (γ = 10.0).

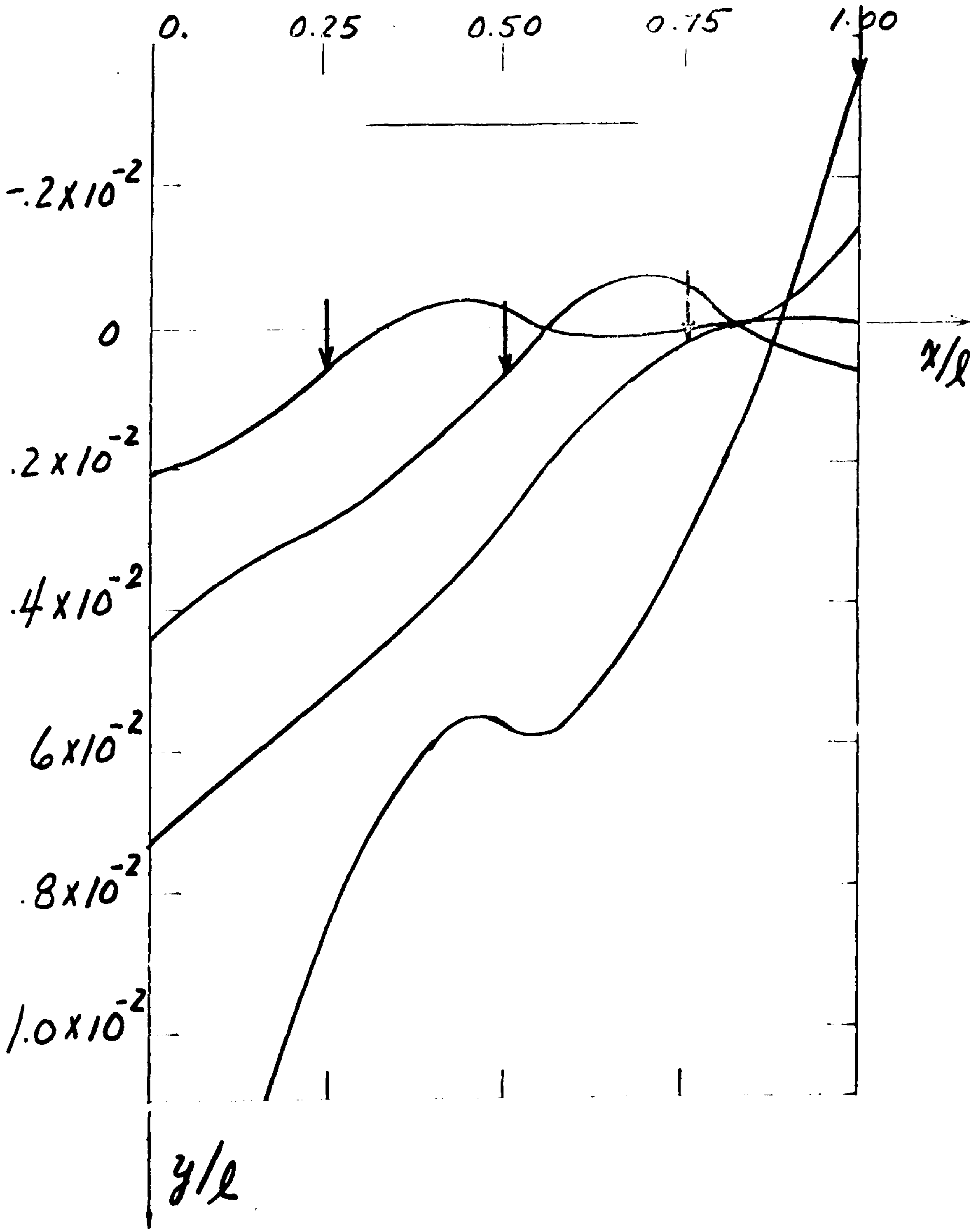

Figure 4. Beam deflections under a moving load for free-free end conditions (γ = 10.0).

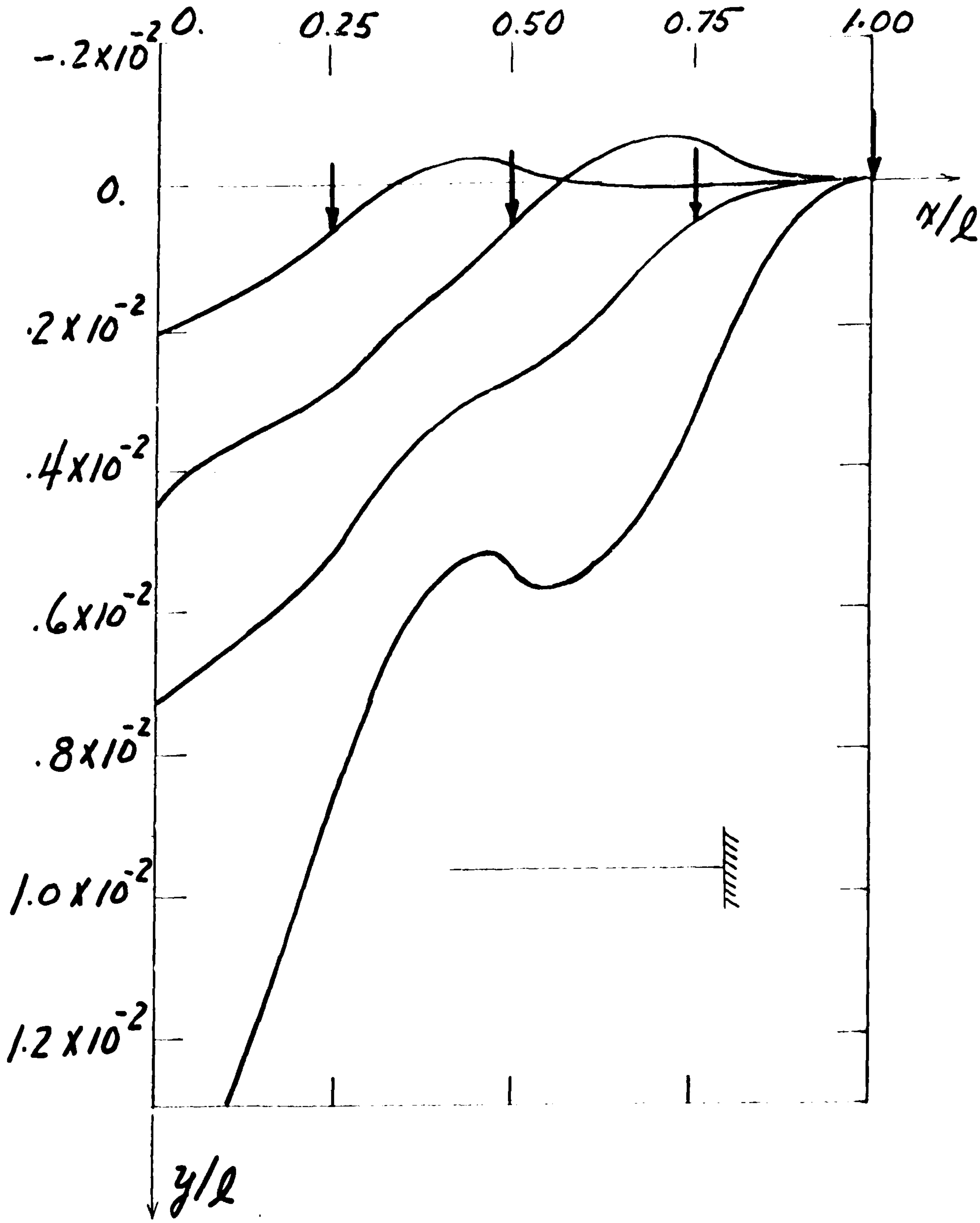

Figure 5. Beam deflections under a moving load for a free-fixed supports (γ = 10.0).

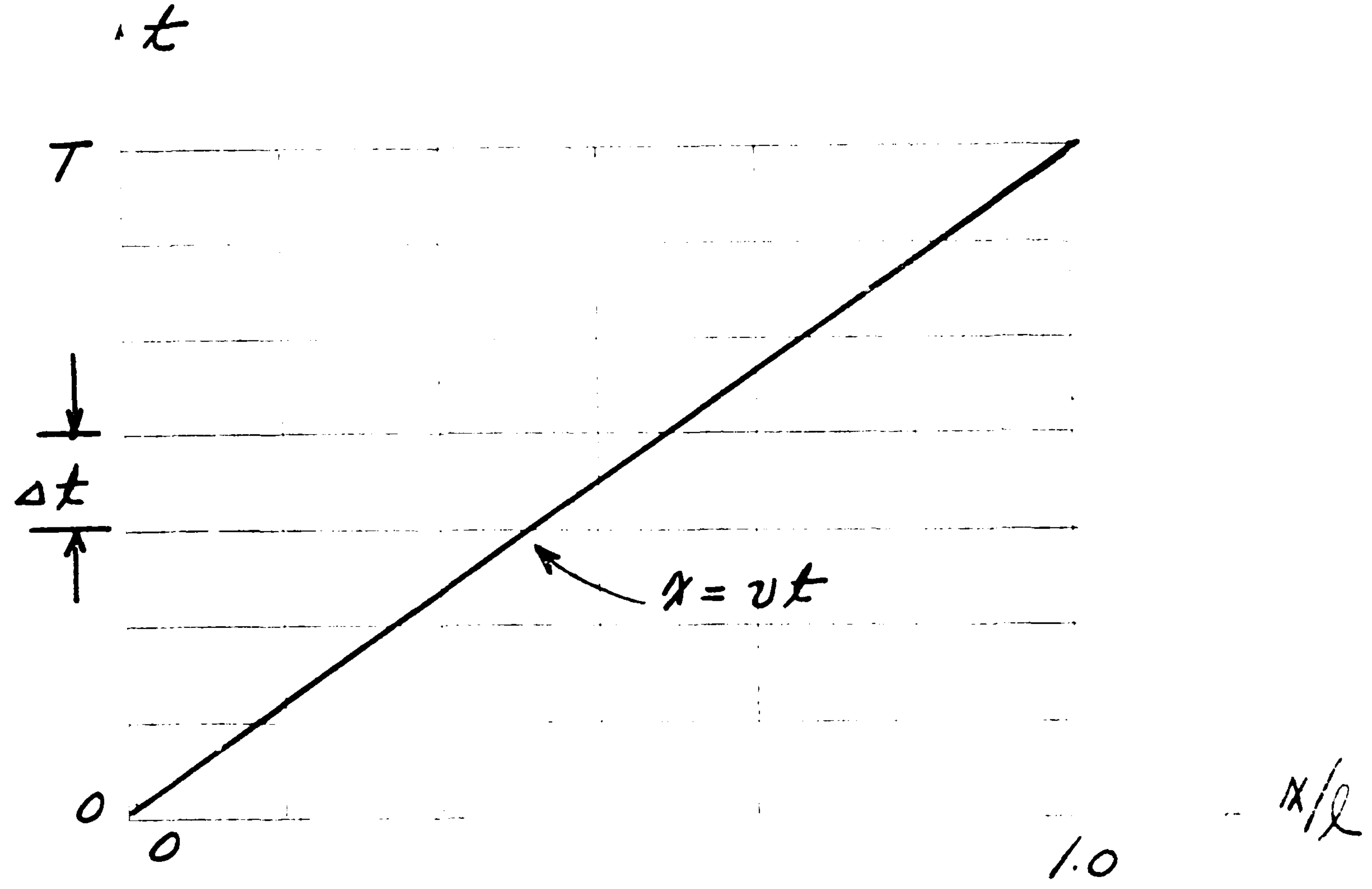

Figure 6

A Typical Grid of Finite Elements

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